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BRUCE HEAVY WATER PLANT
N_2 COMPRESSOR
VIBRATION AND TORQUE ANALYSES

Report No 87-153-K

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ABSTRACT

Experimental vibration and motor shaft torque analyses were carried out to determine the cause(s) of high vibration on the N_2 compressor-motor and the high electrical power requirement of the motor. The investigation revealed that the high vibration on the compressor is due to the proximity of a structural natural frequency to 2 x rpm. In addition, the high motor vibration is most likely due to a rotor natural frequency near 2 x rpm and to electrically defective motor components causing unbalanced magnetic forces at 1 x rpm and 2 x rpm. The motor torque analysis showed that the motor mechanical output power was significantly less than the electrical power draw which also indicated defective electrical components. Inspection of the motor revealed discoloured and frayed stator laminations. Recommendations were made to replace the motor and install an undamped dynamic vibration absorber to detune the compressor structural natural frequency near 2 x rpm.

job	file	date	report no.
27-183-552	825.4	August 18, 1987	87-153-K

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EXECUTIVE SUMMARY

BRUCE HEAVY WATER PLANT
N_2 COMPRESSOR
VIBRATION AND TORQUE ANALYSES

F. Perricone, N.A. Jagan, J.D. Lindsay
Plant Equipment Dynamics Section
Mechanical Research Department

The N_2 compressor-motor system was operating with high vibration and a high motor electrical power draw. Research was requested to investigate these problems to determine the cause(s) and to recommend solutions.

High vibration on the motor was found to be due to:

- a rotor natural frequency near 2 x rpm amplifying rotor harmonic vibration at 2 x rpm
- electrically defective components which produce unbalanced electromagnetic forces causing high vibration at and near 1 x rpm and 2 x rpm.

The motor shaft torque analysis using a strain gauge telemetry system showed that the motor's output power was 27% lower than its rated power. Thus the excessively high electrical power draw indicated electrically defective motor components. Field inspection of the motor revealed discoloured and frayed stator laminations.

High vibrations on the compressor were found to be due to a compressor structural natural frequency near 2 x rpm amplifying the compressor casing vibration at 2 x rpm.

Recommendations were made to install a new motor and an undamped dynamic vibration absorber on the compressor structure.

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To Mr. G. Harris
Bruce Heavy Water Plant

BRUCE HEAVY WATER PLANT N_2 COMPRESSOR VIBRATION ANALYSIS

1.0 INTRODUCTION

Routine vibration measurements carried out by BHWP maintenance personnel indicated that the N_2 compressor-motor system was operating with high vibration levels. In addition, the motor was drawing more power than its rated capacity. BHWP Maintenance requested Mechanical Research to:

1. Measure and determine the severity of the vibration levels.
2. Determine the cause(s) of the high vibrations.
3. Measure the compressor motor's output power.
4. Provide recommendations.

The compressor-motor system consists of an Ingersoll-Rand three stage compressor driven by a General Electric two pole induction motor equipped with journal bearings and rated at 261.1 kW (350 HP) at 3545 rpm.

2.0 MEASUREMENTS AND INSTRUMENTATION

2.1 Measurement Locations

The general compressor-motor arrangement is shown in Figure 1(a). Some of the measurement locations and directions are shown in Figure 1(b).

2.2 Measurements

2.2.1 Overall Velocity Levels

The overall velocity levels were measured using a Bruel & Kjaer Model 4368 accelerometer, a Bruel & Kjaer Model 2635 charge amplifier and a Bently Nevada Digital Vector Filter

(DVF2). Overall velocity levels were obtained while the system was operating under load.

2.2.2 Velocity Spectra

Velocity spectra were generated by analyzing the accelerometer signals (integrated once) with an Ono Sokki CF920 dual channel FFT analyzer. Each velocity spectrum is the result of sixteen averages. Velocity spectra were obtained while the system was operating under load.

2.2.3 Impact Spectra

Structural natural frequencies were determined by impact testing. The compressor and motor structures were excited throughout a range of frequencies (5-200 Hz) with a soft-tipped hammer instrumented with a force transducer. The input force (hammer) and response (accelerometer) signals were fed simultaneously to a dual channel FFT analyzer which was used to calculate the input and response spectra as well as the compliance and inertance transfer functions. The validity of the transfer functions was checked by determining the coherence of the input and response signals. The ambient noise was reduced by averaging the results of eight impacts.

Estimates of rotor natural frequencies were also determined by impact testing on the rotor. Impact testing was carried out when the system was not operating.

2.2.4 Trip Test

A trip test was carried out to assist in determining possible electrical faults in the motor. The rundown signal was taken from a Bruel & Kjaer accelerometer mounted on the motor inboard bearing (horizontal direction), tape recorded and later analyzed using the Ono Sokki CF920 dual channel FFT analyzer.

2.2.5 Motor Output Power

The motor output power was determined directly from motor shaft running torque measurements. Shaft torque was obtained by measuring dynamic strain by means of strain gauges

mounted on the surface of the motor's shaft. Four strain gauges (MM Model CEA-06-187UV-350) were mounted at 45° with respect to the motor shaft axis. These gauges formed a full bridge configuration (Figure 2). Using an FM transmitter attached to the coupling, the strain gauge electrical output signal was transmitted by FM telemetry to an external telemetry receiver (Acurex Corp Model 155). The receiver's output signal was fed into a strip chart recorder (Visicorder - Honeywell Model 1858) to yield strain versus time plots. In addition, the strain signals were analyzed with the dual channel FFT analyzer.

2.3 Instrumentation

The main components of the instrumentation used in obtaining the measurements are listed in Table 1.

3.0 RESULTS

3.1 Overall Velocity Levels

The overall velocity levels are summarized in Tables 2 and 3. The maximum levels measured on the motor and compressor casings are 6.8 mm/s (inboard bearing, horizontal direction) and 6.9 mm/s (2nd stage - inboard end, horizontal direction). These levels plotted on a general machinery severity chart (Figure 13), indicate that the levels are near the Rough severity level. These levels are considered high for this type of rotating machinery.

The velocity spectra taken on the compressor and motor casings (Figures 3-12) indicate that the motor has significant frequency components at 1 x rpm (59.5 Hz) and 2 x rpm (~119 Hz), and that the compressor has a main frequency component at 2 x rpm.

3.2 Dynamic Characteristics

3.2.1 Compressor Structural Natural Frequency

A compliance transfer function spectrum obtained on the compressor inboard structure in the horizontal direction is shown in Figures 14 and 15. It shows a structural natural frequency at 117 Hz.

The high vibration levels at 2 x rpm, measured on the compressor structure, are most likely caused by amplified vibration due to excitation of the lateral structural natural frequency near 2 x rpm.

3.2.2 Motor Rotor Natural Frequency

A compliance transfer function obtained on the motor rotor is shown in Figures 16 and 17. The response was measured near the coupling in the horizontal direction while the motor was shutdown. The impact spectrum shows a rotor natural frequency at 108-109 Hz which is near 2 x rpm (ie, 119 Hz). This rotor critical speed is the primary cause of the high velocity component near 2 x rpm measured on the motor in the horizontal direction (Figure 3).

The test results show that the developed oil film in the journal bearing under operating conditions has a stiffening effect on the rotor and increases the rotor natural frequency closer to 2 x rpm. Figure 18 shows velocity spectra obtained at the motor inboard bearing housing in the horizontal direction at three stages during a trip (rundown) test. Just before the motor is shut off (Figure 18a), the motor operated with a 2 x rpm velocity component of 2.93 mm/s. A fraction of a second after the trip, the 2 x rpm component decreased significantly from 2.93 to 0.968 mm/s and a broad band response centred at 109 Hz* appeared (Figure 18b). Approximately 1 second after the trip, the 2 x rpm component decreased to 0.067 mm/s and the velocity spectrum still displayed the broad band response at 109 Hz (Figure 18c).

In real time**, the shift in the spectral broad band from the 2 x rpm region to 109 Hz can be readily observed in the initial moments of the shutdown.

---

* System natural frequencies often appear as broad band responses in frequency spectral graphs.

**Real time refers to observing the recorded signals played back on a FFT analyzer in the instantaneous FFT mode.

3.3 Motor Electro-Magnetic Vibration

3.3.1 Beating

Time graphs obtained on the motor outboard bearing casing (vertical direction) and the motor inboard bearing casing (horizontal direction) show significant "beating" in the vibration signal (Figures 19 and 20). Both signals have a modulation period of approximately 0.94 seconds which is 2 x slip period. The slip period is defined as follows:

$$T_{slip} = 1/F_{slip}$$

and

$$F_{slip} = F_{line} - F_{running\ speed}$$

where

T_{slip} = Slip Period
F_{slip} = Slip Frequency
F_{line} = Line Frequency
$F_{running\ speed}$ = Running Speed Frequency

This beating in the signal indicates that there is a motor electro-magnetic problem related to improperly installed, or defective rotor and/or stator components.

3.3.2 2 x Slip Frequency Modulation

Zoom velocity spectra (resolution of 0.03125 Hz) obtained on the motor outboard bearing casing (vertical direction) show sideband frequencies at the 1 x rpm (59.469 Hz), the 2 x rpm (118.875 Hz) and the 3 x rpm (178.406 Hz) frequencies spaced at about 1.063 Hz, 1.125 Hz and 1.063 Hz respectively (Figures 21, 22 and 23). These spacing intervals are very close to 2 x slip frequency. Similar zoom velocity spectra obtained on the motor outboard bearing casing (horizontal direction) also show the 1 x rpm and 2 x rpm frequencies with 2 x slip frequency sidebands (Figures 24 and 25). The presence of the sidebands around 1 x and 2 x running speed is due to modulation of the 1 x rpm and 2 x rpm frequencies by the 2 x slip frequency.

3.3.3 Trip Test

A peakhold velocity spectrum, obtained on the motor inboard bearing casing horizontal direction during rundown of the motor, is shown in Figure 26. Figure 27 shows velocity spectra obtained on the motor inboard bearing casing (horizontal direction) just before shutoff (Figure 27a), a fraction of a second after shutoff (Figure 27b), and approximately 1 second after shutoff (Figure 27c). These figures show a sudden significant drop in the vibration amplitude at 1 x rpm which occurred immediately after the power was turned off. This is indicative of electrically defective motor components.

3.4 Motor Power

The motor was running with an electrical power draw which was significantly more than its rated power. In order to assist in determining if the motor was electrically defective, the motor's output shaft was instrumented with strain gauges to measure the shaft mechanical power during normal operating conditions. The strain gauge signals were measured using a telemetry system.

The test results showed that the motor's running torque was 502 N.m (4443 in.lb). The corresponding motor output power was calculated to be 189 kW (254 HP). This is about 27% lower than the motor's rated power. These results indicate that the motor most likely had electrically defective components.

3.5 Field Inspection of the Motor

A field inspection of the motor was carried out and the investigation revealed that:

- the stator bars were discoloured and showed signs of overheating
- the stator laminations were discoloured and frayed.

These results confirmed the experimental vibration and torque analysis which indicated that the motor had electrically defective components.

4.0 CONCLUSIONS

4.1 Motor and compressor vibration are considered high and corrective actions are recommended.

4.2 The main cause of the high motor vibration is a rotor critical speed near 2 x rpm amplifying rotor vibration.

4.3 The main cause of the compressor high vibration is a structural natural frequency near 2 x rpm amplifying compressor structure vibration.

4.4 The analyses indicated that the motor had electrically defective components causing high vibration at and near 1 and 2 x rpm. Inspection of the motor confirmed that the motor had defective components.

5.0 RECOMMENDATIONS

5.1 The motor should be replaced.

5.2 An undamped dynamic vibration absorber should be installed on the compressor casing to detune the compressor structural natural frequency near 2 x rpm.

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TABLE 1

INSTRUMENTATION LIST

1.	Accelerometer:	Bruel & Kjaer Model 4368
2.	Charge Amplifier:	Bruel & Kjaer Model 2635
3.	Tracking Filter:	Bently Nevada DVF 2
4.	Dual Channel Analyzer:	Ono Sokki CF 920
5.	Digital Plotter:	Hewlett-Packard Model 7470A
6.	Oscilloscope:	Textronix Model 5110 MOD 709V
7.	Tape Recorder:	Kyowa Model RTP 600 B
8.	Filter:	Kron-Hite Model 3343
9.	Vibration Calibrator:	General Radio Model 1557-A
10.	Visicorder:	Honeywell Model 1858
11.	Strain Gauge:	Micro-Measurements Div. Model CEA-06-187UV-350
12.	Telemetry Receiver:	Acurex Corp. Model 155
13.	Impact Hammer:	Instrumented Tip Hammer PCB Piezotronics Inc, Model 086B50

TABLE NO. 2

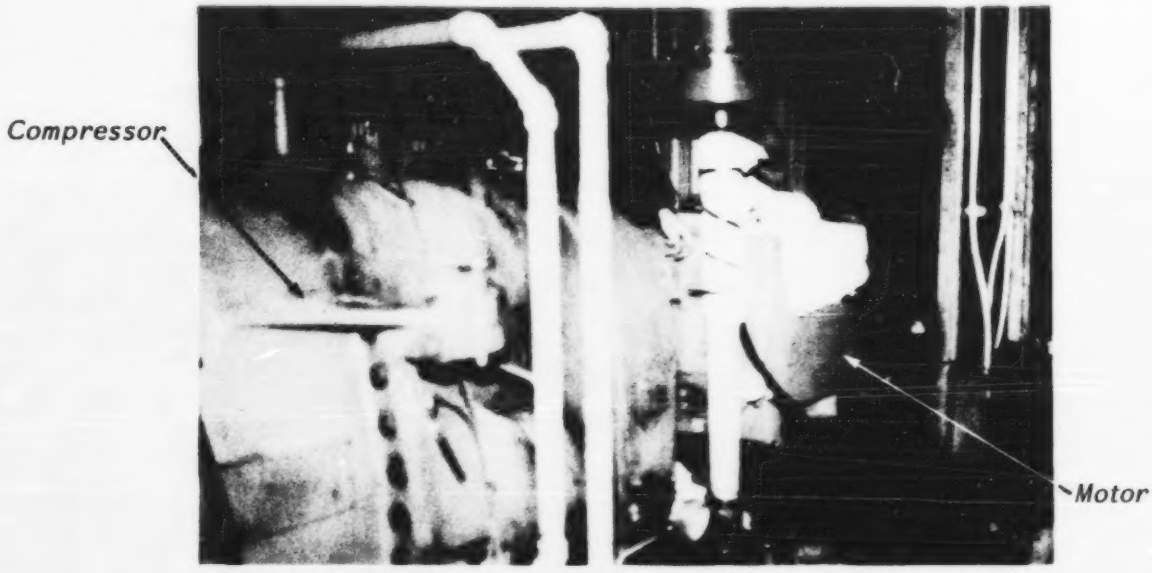
MOTOR BEARING HOUSING OVERALL VELOCITY LEVELS

Measurement Location	Velocity Level mm/s	Figure No.
Motor OB Horiz	3.3 - 4.1	3
Motor OB Vert	2.3 - 3.8	4
Motor OB Axial	3.3 - 4.6	5
Motor IB Horiz	5.7 - 6.8	6
Motor IB Vert	3.0 - 3.6	7
Motor IB Axial	3.9 - 5.3	8

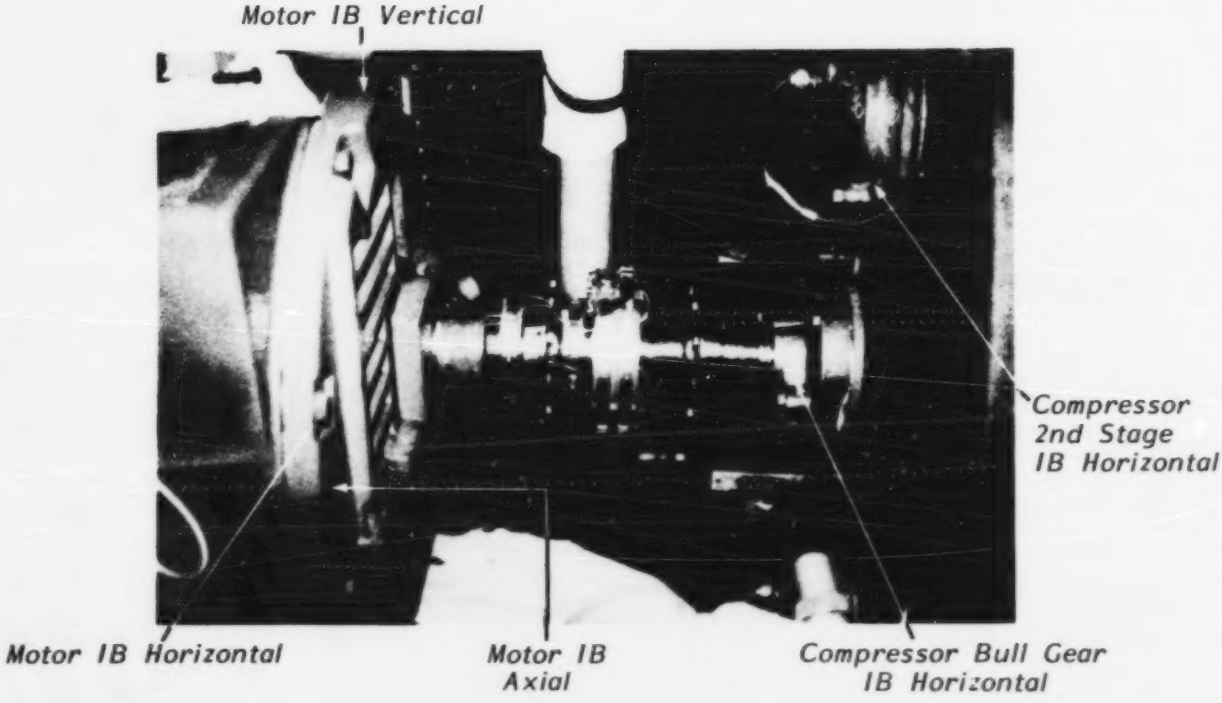
TABLE NO. 3

COMPRESSOR STRUCTURE OVERALL VELOCITY LEVELS

Measurement Location	Overall Velocity mm/s	Figure No.
1st Stage Compressor - IB Horiz	4.5 - 5.2	9
2nd Stage Compressor - IB Horiz	5.2 - 6.9	10
3rd Stage Compressor - IB Horiz	3.8 - 6.0	11
Bull Gear - IB Vert	2.9 - 3.3	12



(a) N_2 Compressor - Motor System



(b) Measurement Locations

FIGURE 1

174046-RD

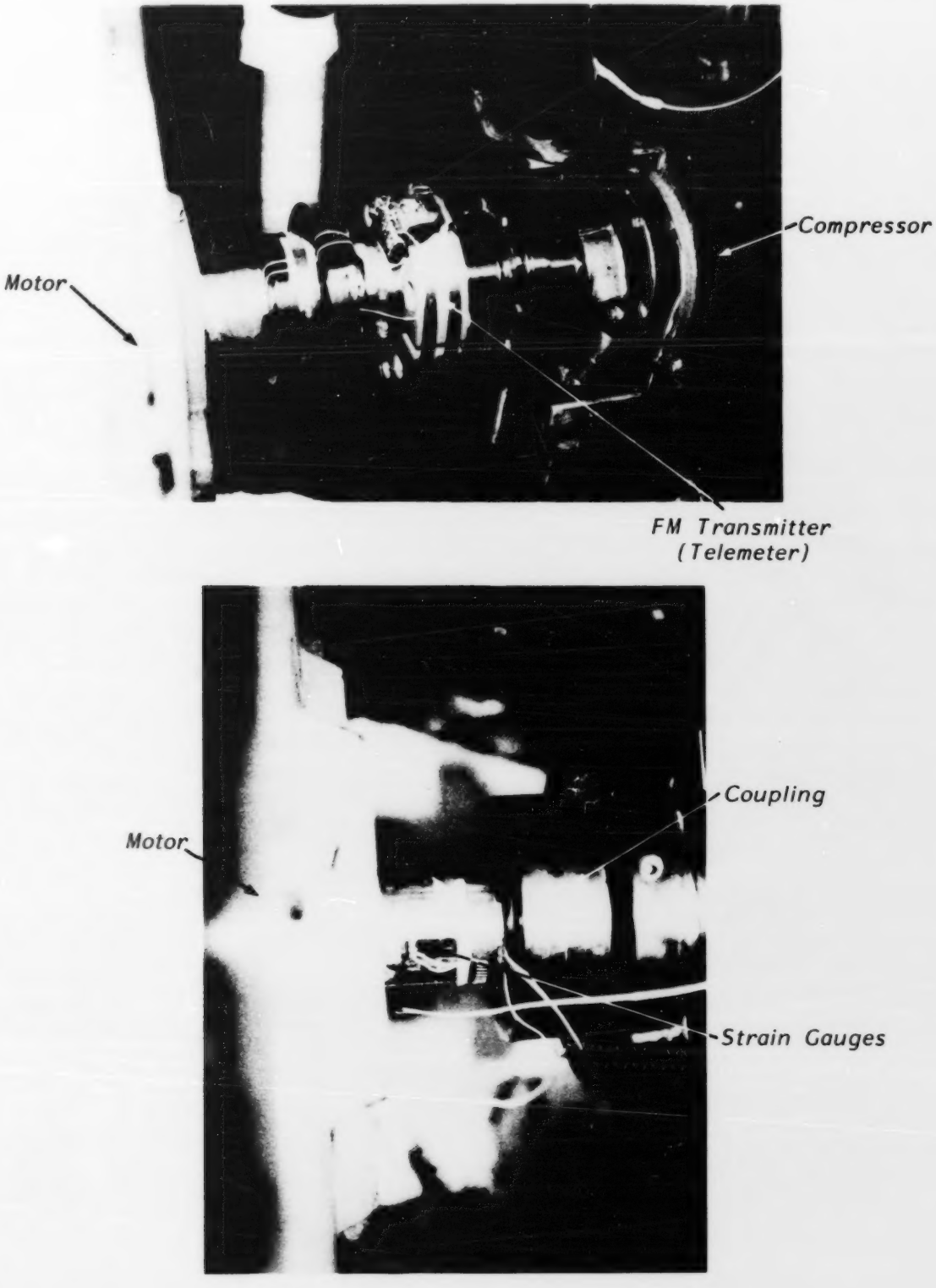


FIGURE 2

STRAIN GAUGE ASSEMBLY ON MOTOR SHAFT

174047-RD

BHWP N2 COMP - MOTOR OB HORIZ VEL SPECTRUM
200Hz A: AC/ 50V B: AC/ 50V INST 0/16 ChA 2k

	PWR SPECTRUM	ChA	
1	59.500Hz	.640E+0	EU (mm/s)
2	119.000	.208E+1	
3	178.500	.149E+0	
4			
5			
6			
7			
8			
9			
10			

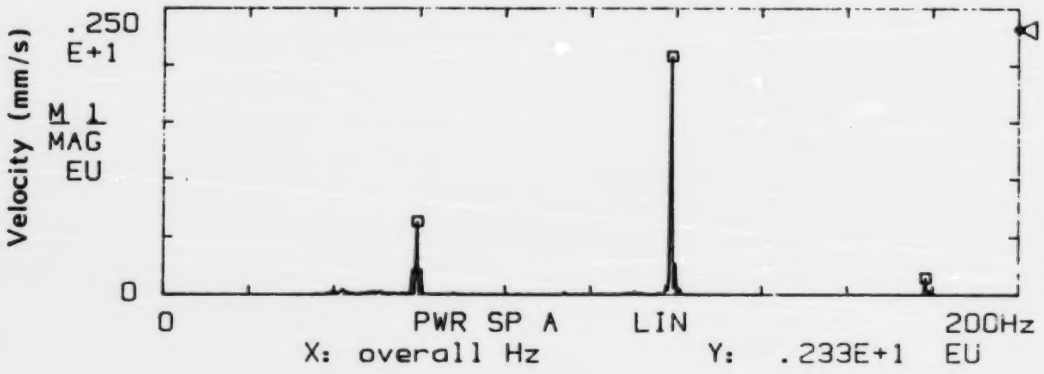


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FIGURE 3

MOTOR OUTBOARD BEARING CASING VELOCITY HORIZONTAL DIRECTION

BHWP N2 COMP - MOTOR OB VERT VEL SPECTRUM
200Hz A: AC/ 50V B: AC/ 50V INST 0/16 ChA 2k

	PWR SPECTRUM	ChA	
1	59.500Hz	.878E+0	EU (mm/s)
2	119.000	.897E+0	
3	178.500	.258E+0	
4			
5			
6			
7			
8			
9			
10			

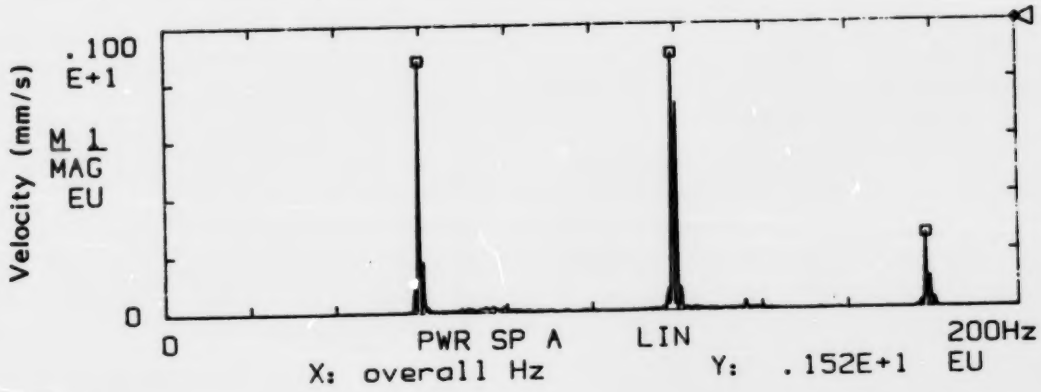

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FIGURE 4

MOTOR OUTBOARD BEARING CASING VELOCITY VERTICAL DIRECTION

BHWP N2 COMP - MOTOR OB AXIAL VEL SPECTRUM
200Hz A: AC/ 50V B: AC/ 50V INST 0/16 ChA 2k

	PWR SPECTRUM	ChA	
1	59.500Hz	.115E+1	EU (mm/s)
2	59.750	.463E+0	
3	119.000	.697E+0	
4	178.500	.246E+0	
5			
6			
7			
8			
9			
10			

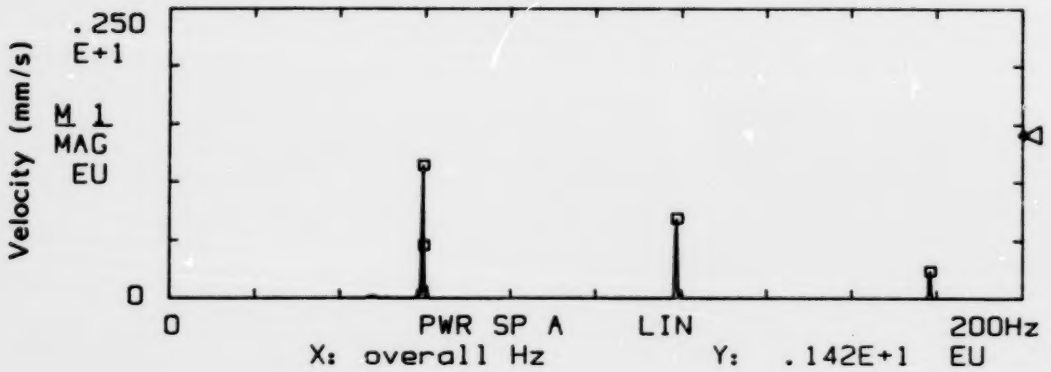

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FIGURE 5

MOTOR OUTBOARD BEARING CASING VELOCITY AXIAL DIRECTION

BHWP N2 COMP - MOTOR IB HORIZ VEL SPECTRUM
200Hz A: AC/ 50V B: AC/ 50V INST 0/16 ChA 2k

	PWR SPECTRUM	ChA	
1	59.500Hz	.127E+1	EU (mm/s)
2	118.750	.234E+1	
3	120.000	.551E+0	
4	178.250	.228E+0	
5			
6			
7			
8			
9			
10			

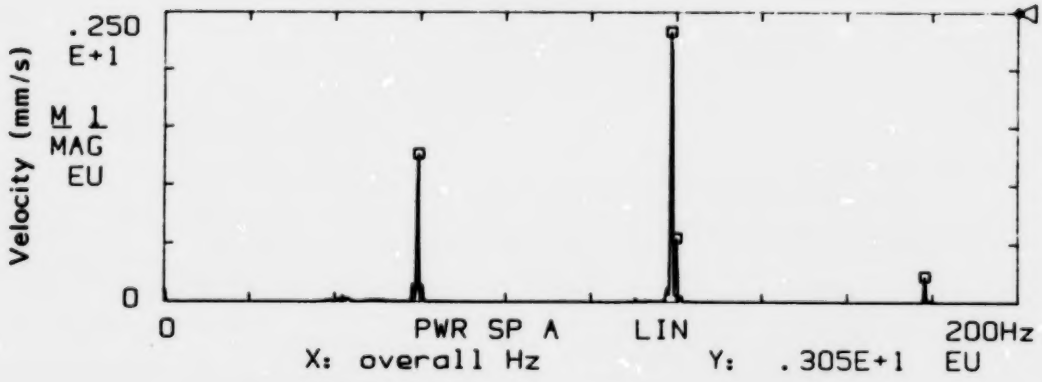

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FIGURE 6

MOTOR INBOARD BEARING CASING VELOCITY HORIZONTAL DIRECTION

BHWP N2 COMP - MOTOR IB VERT VEL SPECTRUM
200Hz A: AC/ 50V B: AC/ 50V INST 0/16 ChA 2k

	PWR SPECTRUM	ChA	
1	59.500Hz	.697E+0	EU (mm/s)
2	119.000	.729E+0	
3	120.000	.327E+0	
4			
5			
6			
7			
8			
9			
10			

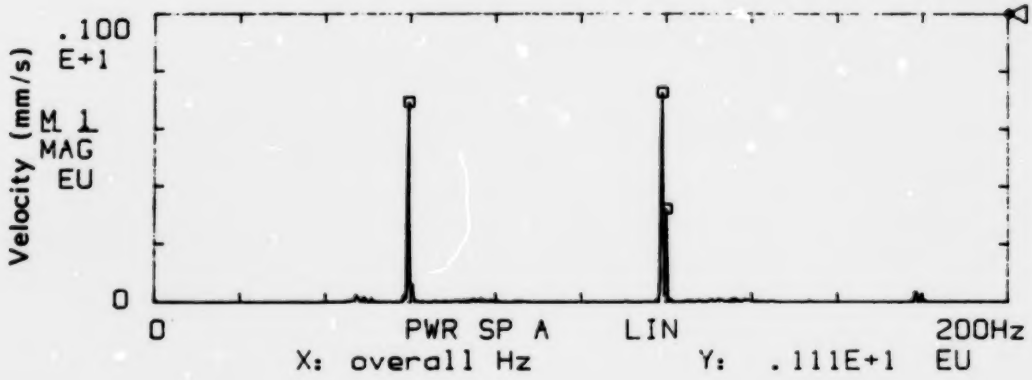

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FIGURE 7

MOTOR INBOARD BEARING CASING VELOCITY VERTICAL DIRECTION

BHWP N2 COMP - MOTOR IB AXIAL VEL SPECTRUM
200Hz A: AC/ 50V B: AC/ 50V INST 0/16 ChA 2k

	PWR SPECTRUM	ChA	
1	59.500Hz	.968E+0	EU (mm/s)
2	119.000	.217E+1	
3	119.250	.711E+0	
4	178.500	.413E+0	
5			
6			
7			
8			
9			
10			

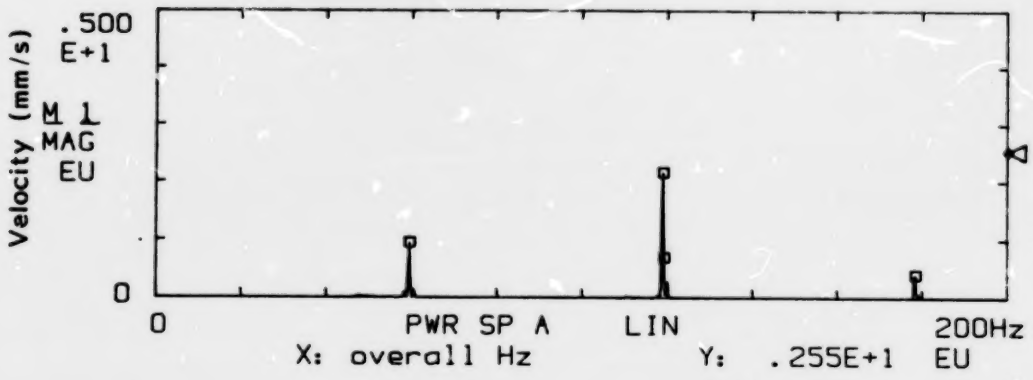

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FIGURE 8

MOTOR INBOARD BEARING CASING VELOCITY AXIAL DIRECTION

BHWP N2 COMP - COMP 1ST STAGE IB HORIZ VEL SPECTRUM
200Hz A: AC/ 50V B: AC/ 50V INST 0/16 ChA 2k

	PWR SPECTRUM	ChA	
1	59.375Hz	.321E+0	EU (mm/s)
2	118.750	.150E+1	
3	178.125	.921E+0	
4	238.125	.347E+0	
5			
6			
7			
8			
9			
10			

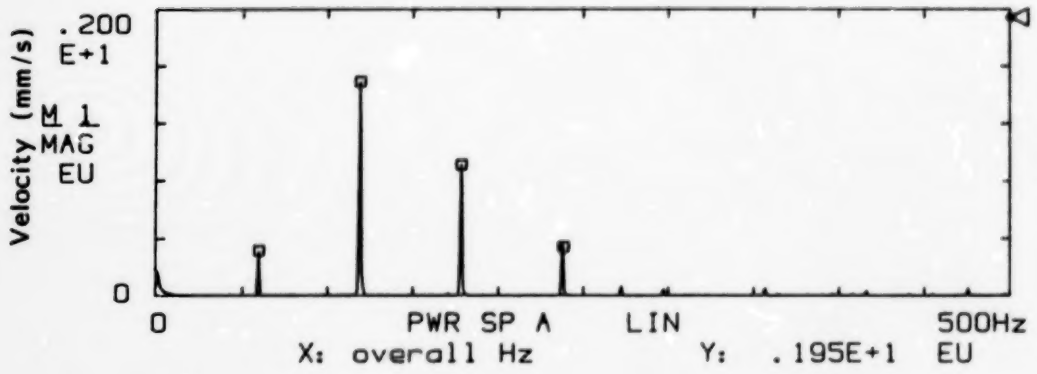

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FIGURE 9

COMPRESSOR 1st STAGE CASING - INBOARD END VELOCITY HORIZONTAL DIRECTION

BHWP N2 COMP - COMP 2ND STAGE IB HORIZ VEL SPECTRUM
200Hz A: AC/ 50V B: AC/ 50V INST 0/16 ChA 2k

	PWR SPECTRUM	ChA	
1	59.375Hz	.456E+0	EU (mm/s)
2	118.750	.387E+1	
3	178.125	.642E+0	
4	238.125	.957E-1	
5			
6			
7			
8			
9			
10			

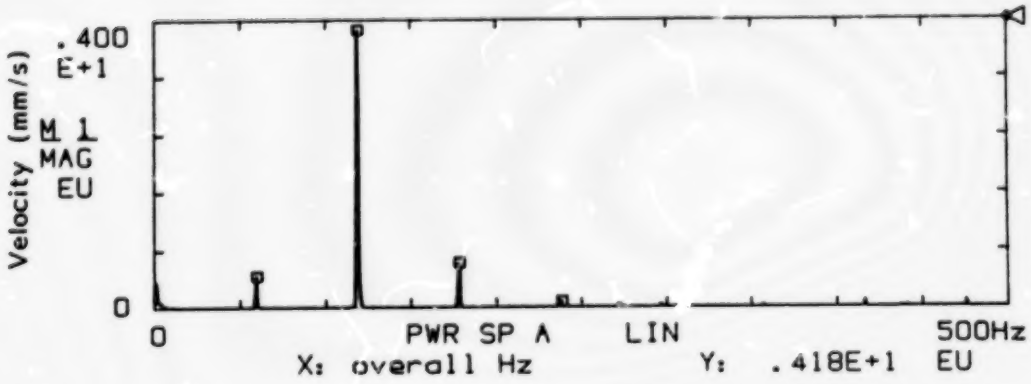

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FIGURE 10

COMPRESSOR 2nd STAGE CASING - INBOARD END VELOCITY
HORIZONTAL DIRECTION

BHWP N2 COMP - COMP 3RD STAGE IB HORIZ VEL SPECTRUM
200Hz A: AC/ 50V B: AC/ 50V INST 0/16 ChA 2k

	PWR SPECTRUM	ChA	
1	60.00Hz	.513E+0	EU (mm/s)
2	118.75	.190E+1	
3	178.75	.772E+0	
4	237.50	.244E+0	
5	535.00	.216E+0	
6	633.75	.338E+0	
7	735.00	.985E+0	
8	875.00	.321E+0	
9			
10			

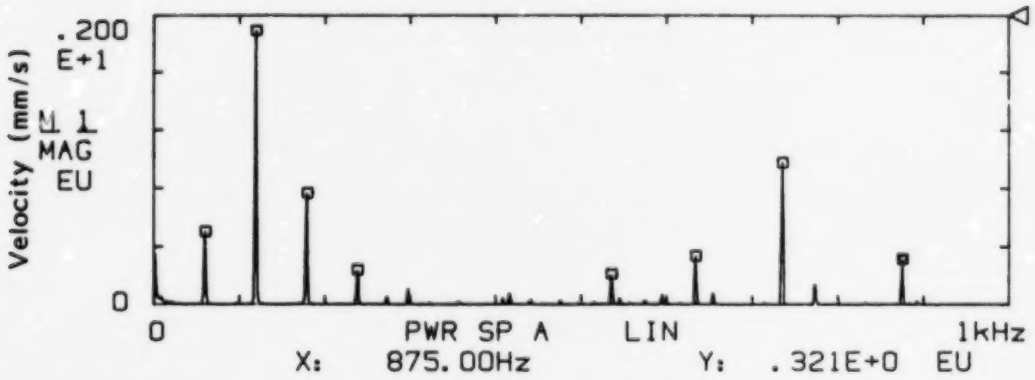

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FIGURE 11

COMPRESSOR 3rd STAGE CASING - INBOARD END VELOCITY
HORIZONTAL DIRECTION

BHWP N2 COMP - COMP BULL GEAR IB HORIZ VEL SPECTRUM
200Hz A: AC/ 50V B: AC/ 50V INST 0/16 ChA 2k

	PWR SPECTRUM	ChA	
1	59.375Hz	.520E+0	EU (mm/s)
2	118.750	.363E+0	
3	178.125	.123E+1	
4	238.125	.133E+0	
5	296.875	.291E+0	
6			
7			
8			
9			
10			

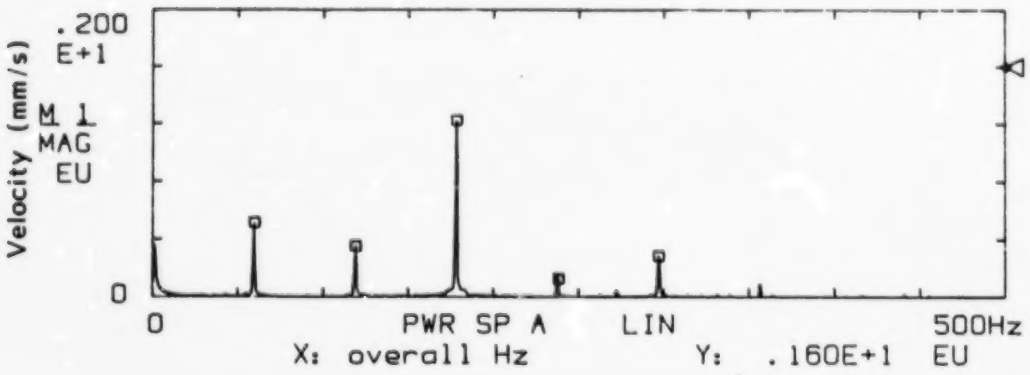

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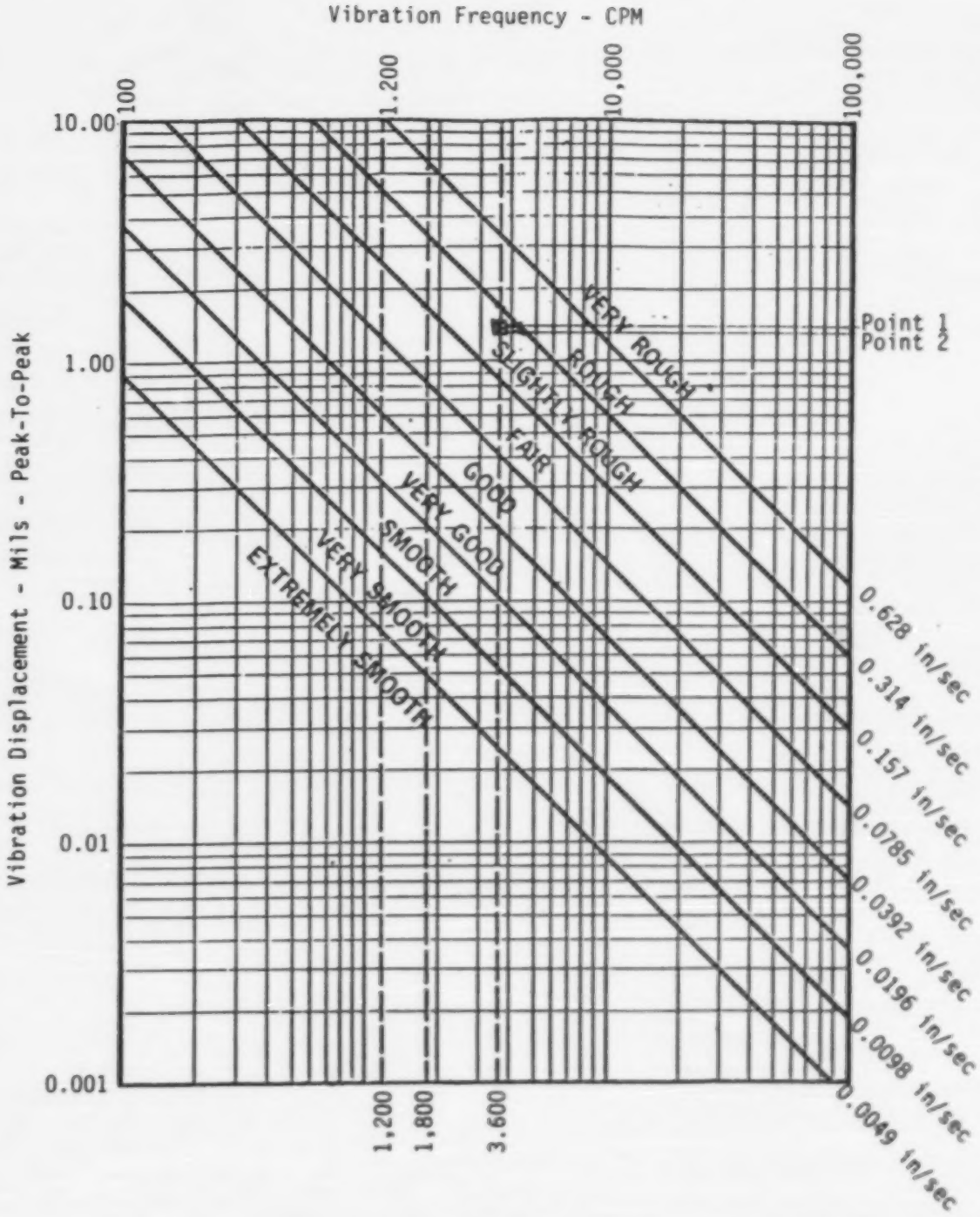
FIGURE 12

COMPRESSOR CASING NEAR BULL GEAR - INBOARD END VELOCITY
HORIZONTAL DIRECTION



Point 1 - Maximum Velocity Level Measured on the Compressor
Point 2 - Maximum Velocity Level Measured on the Motor

FIGURE 13

MAXIMUM MOTOR AND COMPRESSOR VELOCITY LEVELS PLOTTED ON A GENERAL MACHINERY VIBRATION SEVERITY CHART

BHWP N2 COMP - COMP STRUCTURE IB HORIZ IMPACT SPEC
200Hz A: AC/ 50V B: AC/ 50V INST 0/16 DUAL 1k

		XFER FUNC MAG - PH	
1	42.00Hz	.201E-8 EU/EU	162.7 deg
2	67.50	.686E-8	97.2
3	69.00	.501E-8	121.1
4	107.50	.409E-8	162.2
5	111.00	.757E-8	147.8
6	113.50	.795E-8	138.7
7	117.00	.154E-7	84.8
8	120.00	.159E-7	57.8
9	125.00	.525E-8	63.1
10	175.50	.297E-8	65.7

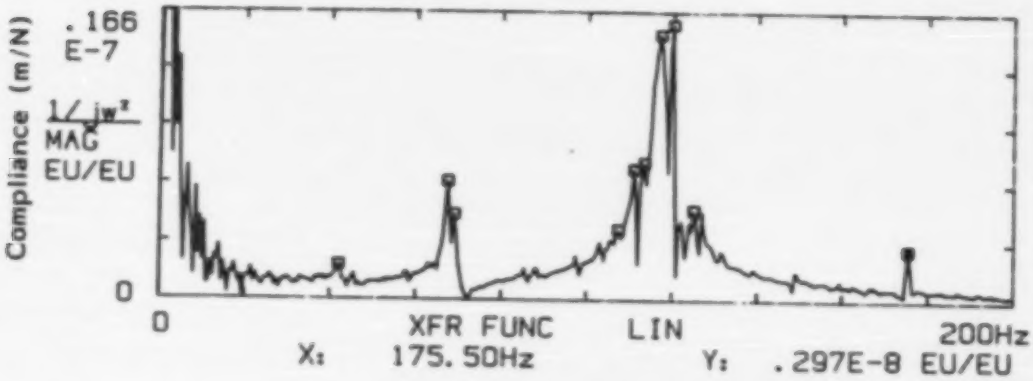


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FIGURE 14

COMPRESSOR CASING - INBOARD END COMPLIANCE TRANSFER FUNCTION
HORIZONTAL DIRECTION

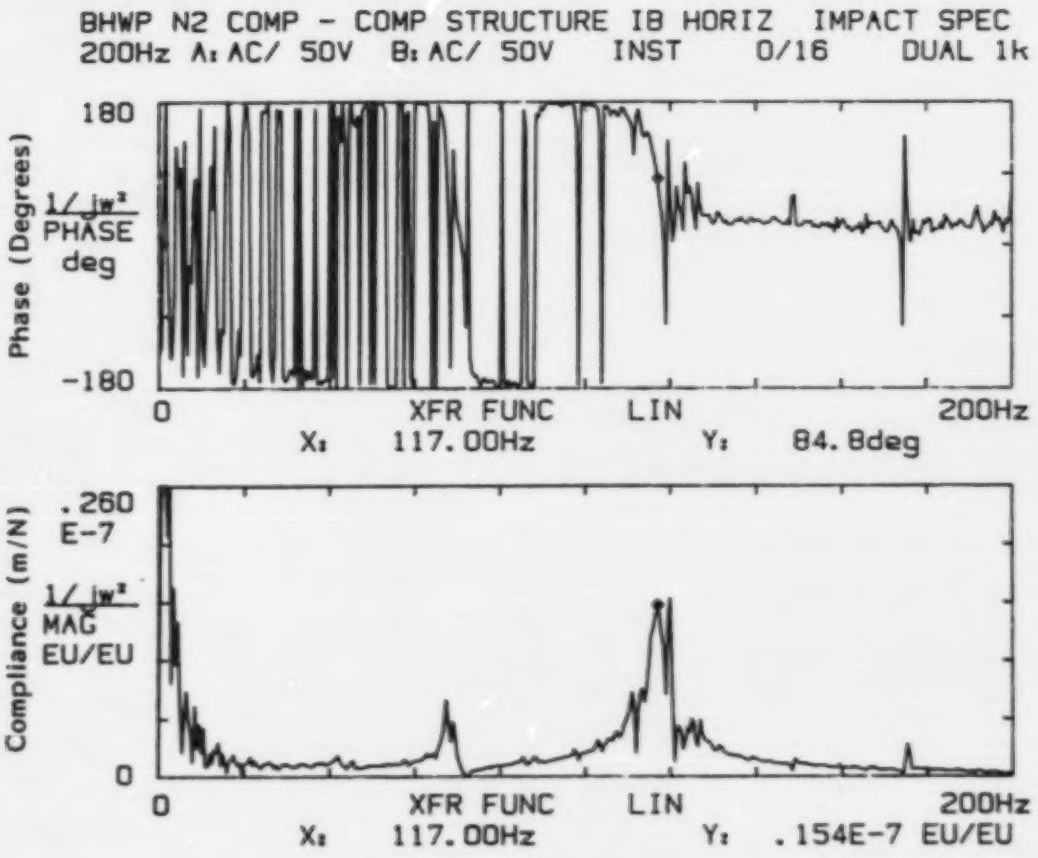

FIGURE 15

COMPRESSOR CASING - INBOARD END COMPLIANCE TRANSFER FUNCTION
HORIZONTAL DIRECTION

BHWP N2 COMP - ROTOR IB HORIZ IMPACT SPECTRUM
200Hz A: AC/ 50V B: AC/ 50V INST 0/16 DUAL 1k

		XFER FUNC MAG - PH	
1	41.50Hz	.102E-6 EU/EU	-106.2 deg
2	49.00	.493E-7	-151.1
3	108.00	.707E-7	-116.1
4	124.00	.116E-7	-168.3
5	134.00	.555E-8	168.5
6	171.50	.430E-8	178.6
7			
8			
9			
10			

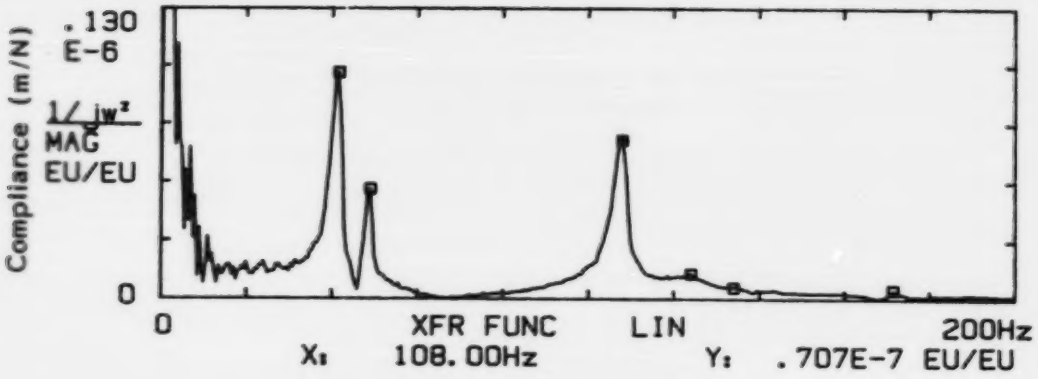

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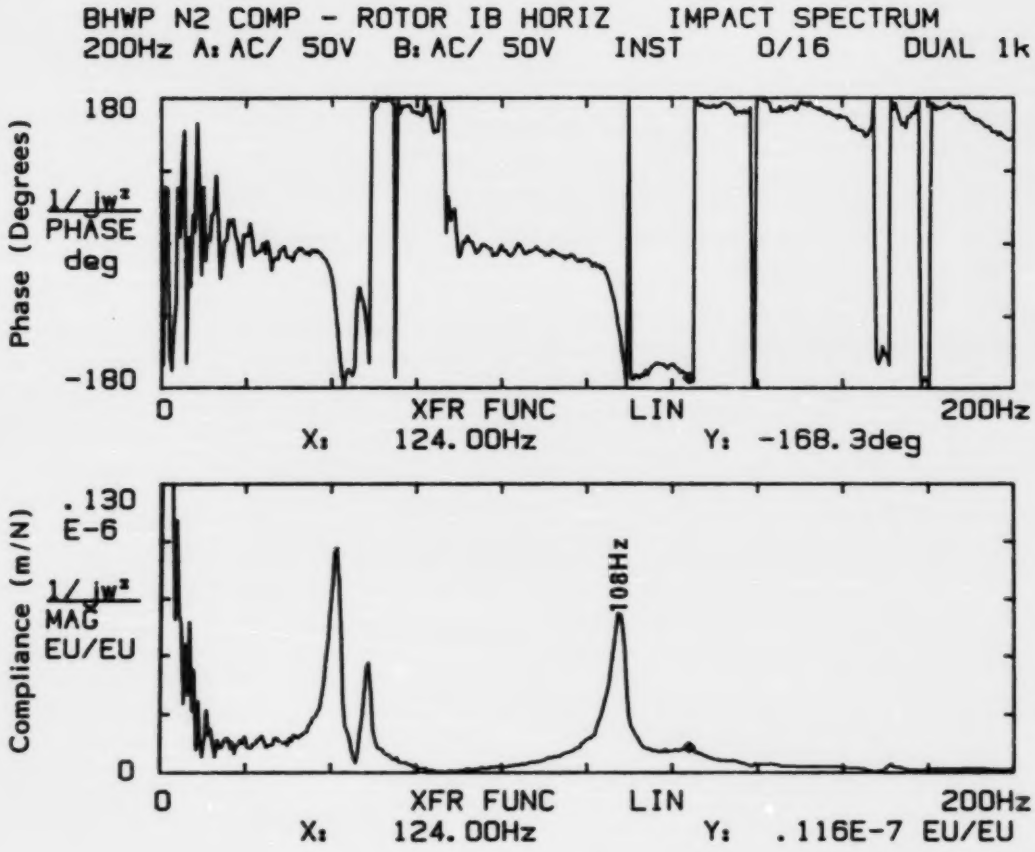
FIGURE 16

MOTOR ROTOR (NEAR COUPLING) COMPLIANCE TRANSFER FUNCTION
HORIZONTAL DIRECTION



00: 43

FIGURE 17

MOTOR ROTOR (NEAR COUPLING) COMPLIANCE TRANSFER FUNCTION
HORIZONTAL DIRECTION

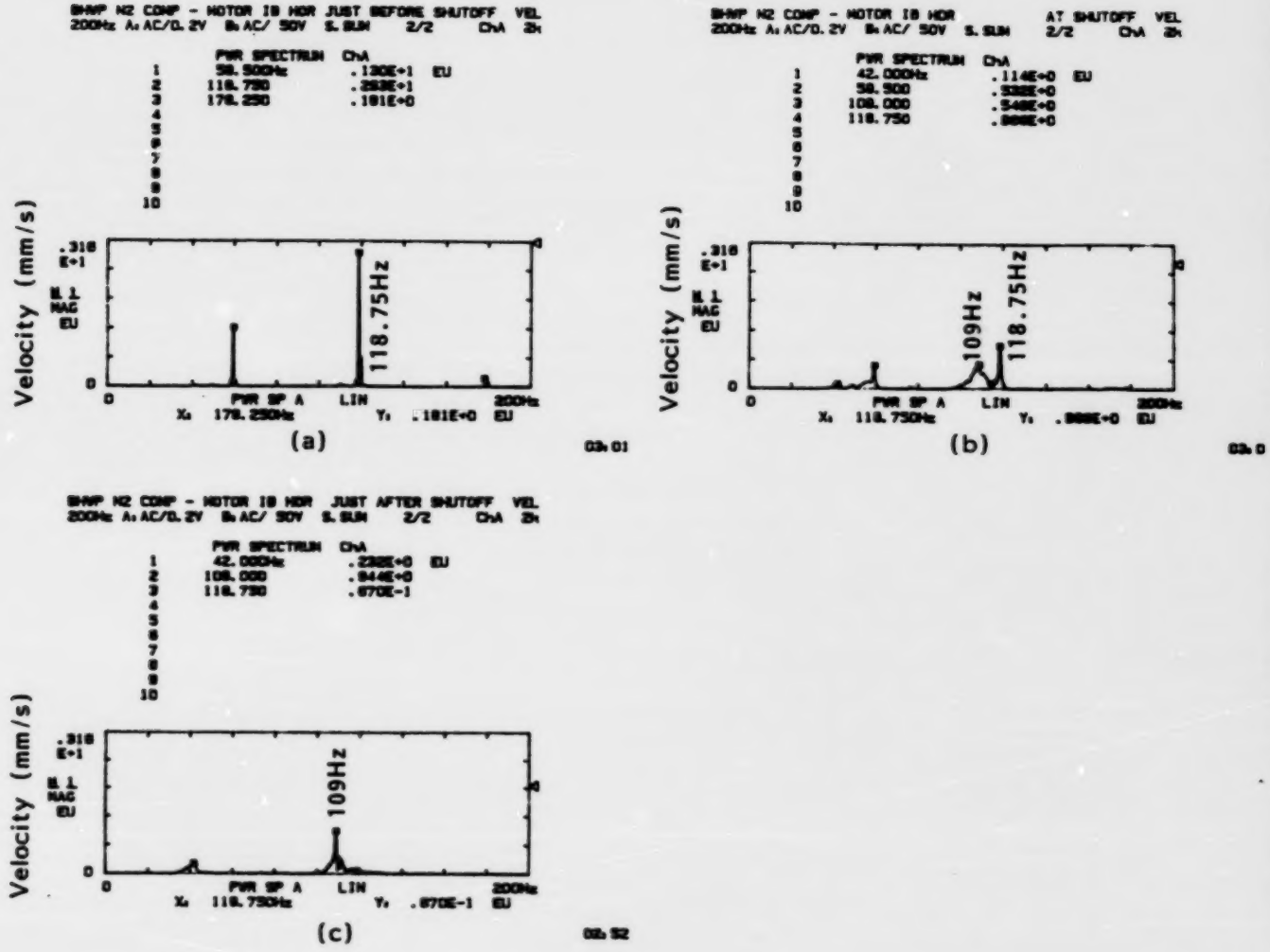


FIGURE 18

MOTOR INBOARD BEARING CASING VELOCITY HORIZONTAL DIRECTION
- RUNDOWN TEST

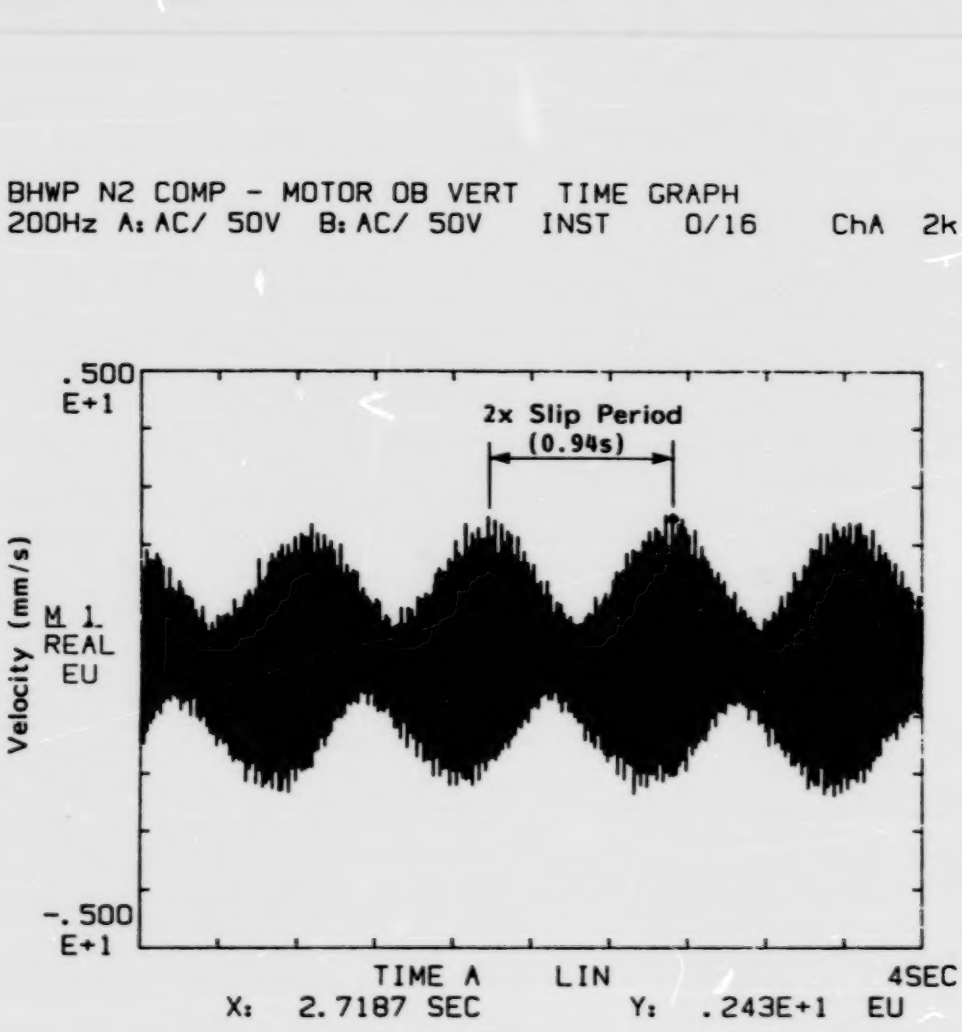


FIGURE 19

MOTOR OUTBOARD BEARING CASING VELOCITY VERTICAL DIRECTION - TIME GRAPH

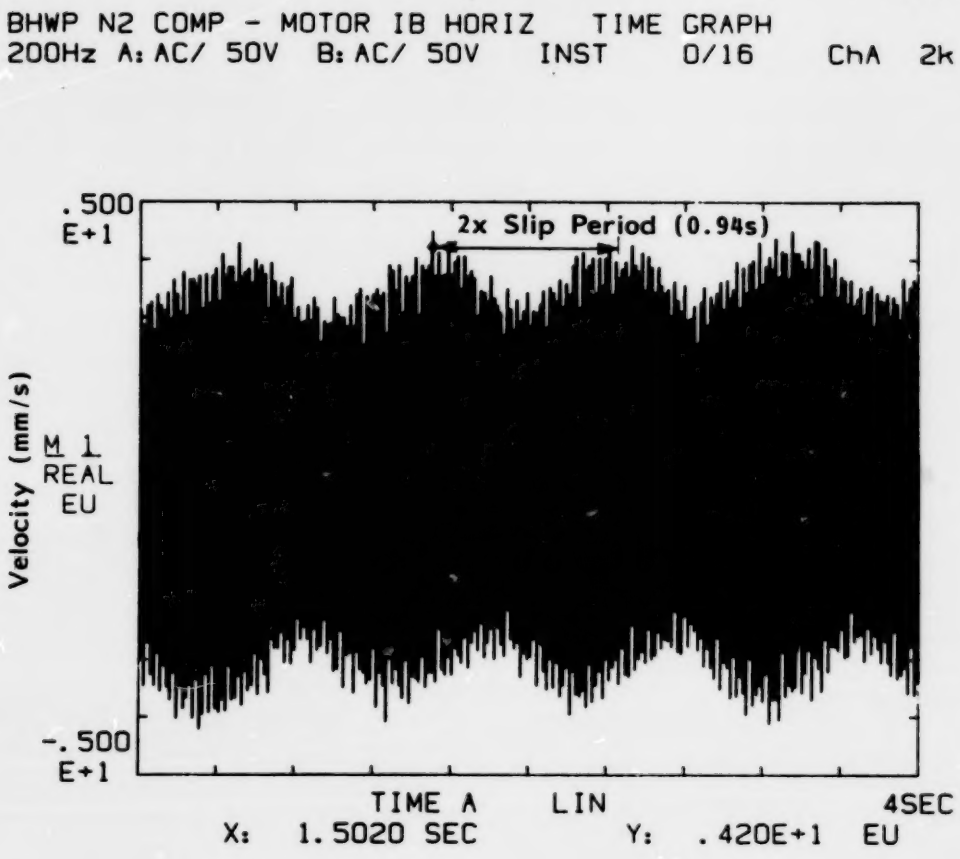


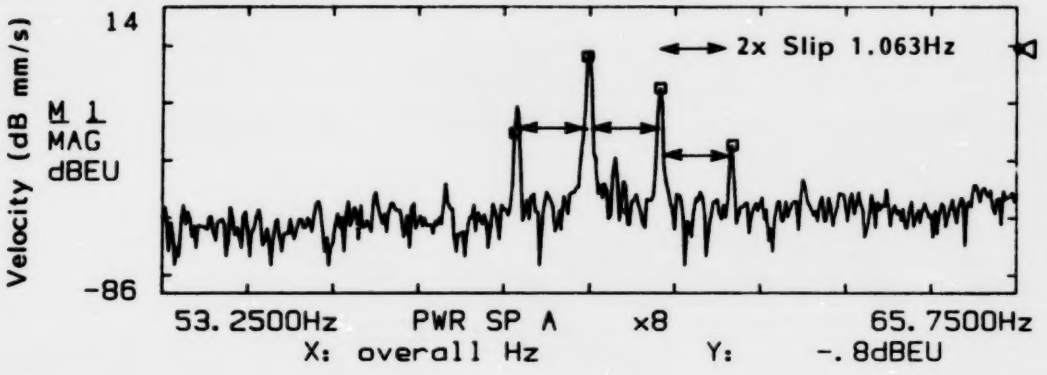
FIGURE 20

MOTOR INBOARD BEARING CASING VELOCITY HORIZONTAL DIRECTION - TIME GRAPH

BHWP N2 COMP - MOTOR OB VERT VEL SPEC ZOOM 60 HZ
200Hz A: AC/ 50V B: AC/ 50V INST 0/16 ChA 2k

	PWR SPECTRUM	ChA
1	58.4062Hz	-30.6dBEU*
2	59.4687	-3.6 (0.66mm/s)
3	60.5312	-14.9
4	61.5937	-34.5
5		
6		
7		
8		
9		
10		

* Reference Velocity is 1mm/s



01:20

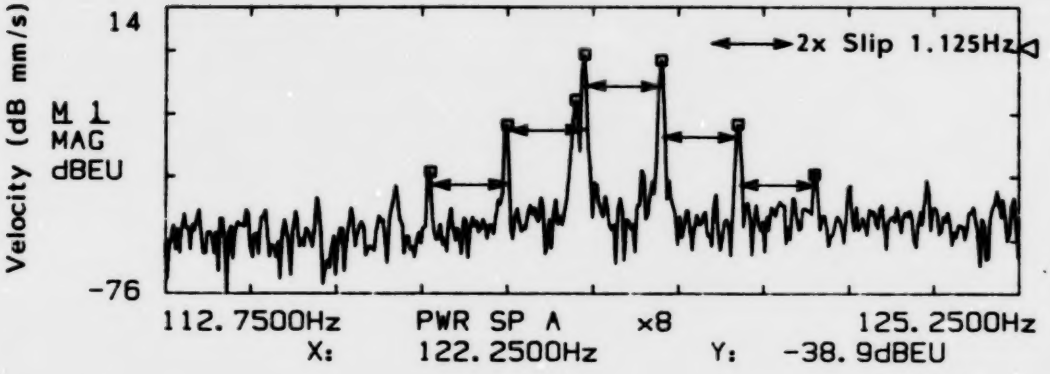
FIGURE 21

MOTOR OUTBOARD BEARING CASING VELOCITY VERTICAL DIRECTION - ZOOM SPECTRUM

BHWP N2 COMP - MOTOR OB VERT VEL SPEC ZOOM 120 HZ
200Hz A: AC/ 50V B: AC/ 50V INST 0/16 ChA 2k

	PWR SPECTRUM	ChA
1	116.6250Hz	-38.0dBEU*
2	117.7500	-23.3
3	118.7500	-15.3
4	118.8750	-1.1 (0.88mm/s)
5	120.0000	-3.0
6	121.1250	-22.8
7	122.2500	-38.9
8		
9		
10		

* Reference Velocity is 1mm/s



01:45

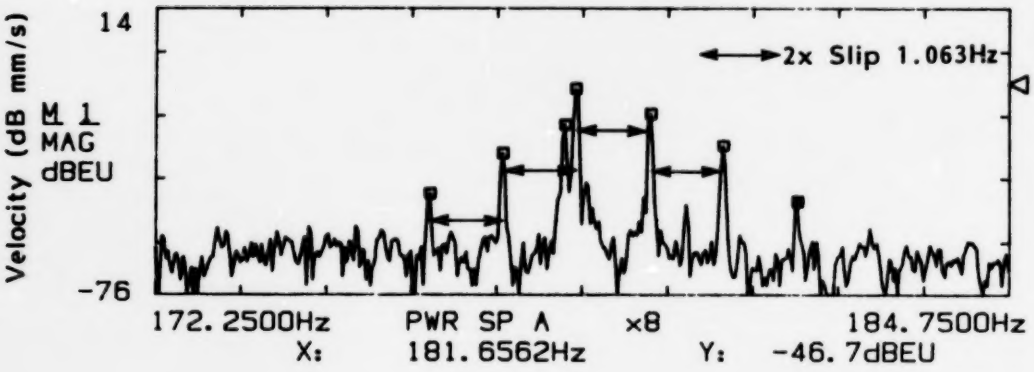
FIGURE 22

MOTOR OUTBOARD CASING VELOCITY VERTICAL DIRECTION - ZOOM SPECTRUM

BHWP N2 COMP - MOTOR OB VERT VEL SPEC ZOOM 180 HZ
200Hz A: AC/ 50V B: AC/ 50V INST 0/16 ChA 2k

	PWR SPECTRUM	ChA
1	176.2812Hz	-44.4dBEU*
2	177.3437	-31.6
3	178.2500	-22.5
4	178.4062	-11.5 (0.27mm/s)
5	179.5000	-19.4
6	180.5625	-29.0
7	181.6562	-46.7
8		
9		
10		

* Reference Velocity is 1mm/s



02: 02

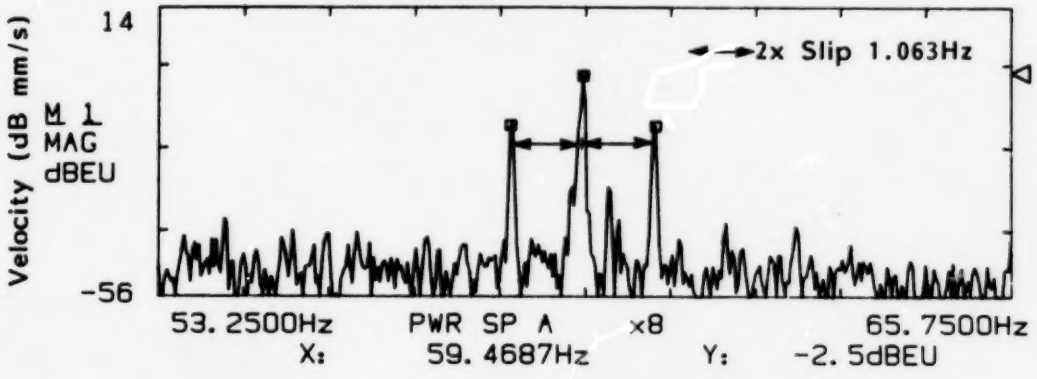
FIGURE 23

MOTOR OUTBOARD BEARING CASING VELOCITY VERTICAL DIRECTION - ZOOM SPECTRUM

BHWP N2 COMP - MOTOR OB HORIZ VEL SPEC ZOOM 60 HZ
200Hz A: AC/ 50V B: AC/ 50V INST 0/16 ChA 2k

	PWR SPECTRUM	ChA
1	58.4062Hz	-14.5dBEU *
2	59.4687	-2.5 (0.75mm/s)
3	60.5312	-14.6
4		
5		
6		
7		
8		
9		
10		

* Reference Velocity is 1mm/s



01:30

FIGURE 24

MOTOR OUTBOARD BEARING CASING VELOCITY HORIZONTAL DIRECTION - ZOOM SPECTRUM

BHWP N2 COMP - MOTOR OB HORIZ VEL SPEC ZOOM 120 HZ
200Hz A: AC/ 50V B: AC/ 50V INST 0/16 ChA 2k

	PWR SPECTRUM	ChA
1	116.8437Hz	-43.3dBEU*
2	117.9062	-20.5
3	118.9687	6.4 (2.09mm/s)
4	120.0312	-11.8
5	121.0937	-24.9
6		
7		
8		
9		
10		

* Reference Velocity is 1mm/s

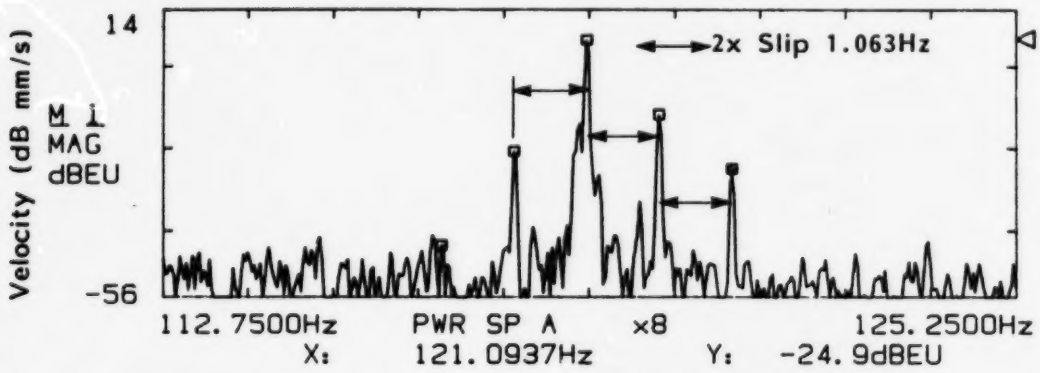


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FIGURE 25

MOTOR OUTBOARD BEARING CASING VELOCITY HORIZONTAL DIRECTION - ZOOM SPECTRUM

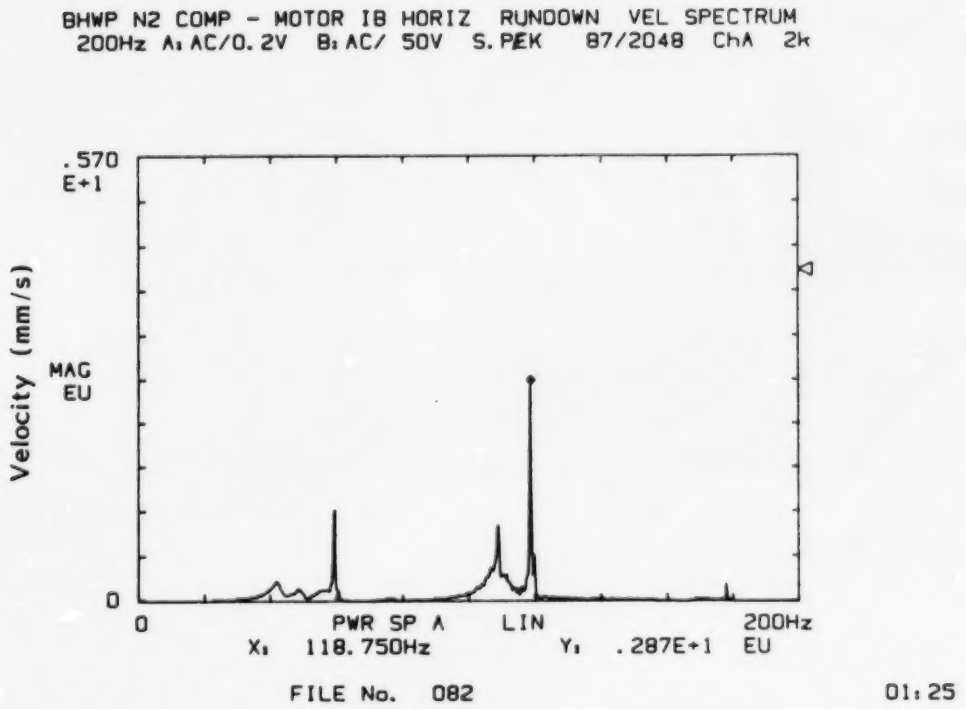

FIGURE 26

MOTOR INBOARD BEARING CASING VELOCITY HORIZONTAL DIRECTION
- PEAK HOLD RUNDOWN SPECTRUM

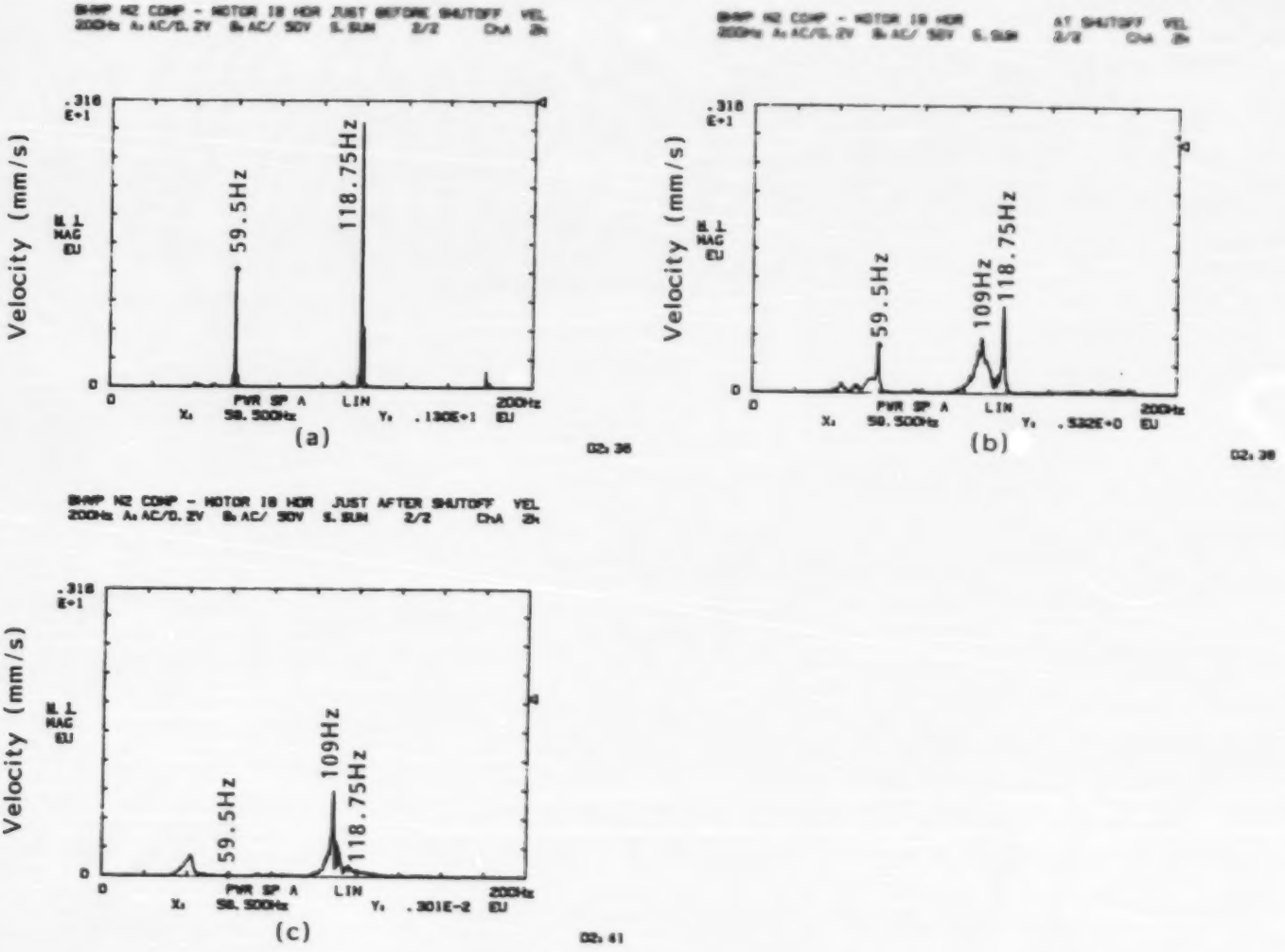

FIGURE 27

MOTOR INBOARD BEARING CASING VELOCITY HORIZONTAL DIRECTION
- RUNDOWN TEST

APPENDIX A

SHAFT TORQUE AND OUTPUT POWER MEASUREMENTS

For a solid round shaft in pure torsion, the shear stress is zero at the centre and maximum at the surface. The shear stress at the surface of the shaft is given by:

$$S = \frac{Tr}{J} \qquad (1)$$

where

T = Torque, N.m
S = Shear stress, Pa
r = Shaft radius, m
J = Polar moment of inertia, m^4

For a solid circular shaft:

$$J = \frac{\pi d^4}{32} \qquad (2)$$

where

d = shaft diameter, m

Substituting (2) into (1) and rearranging yields:

$$T = \frac{\pi S d^3}{16} \qquad (3)$$

The maximum tensile and compressive strains of a shaft in torsion lie on the planes at 45° with respect to the axis. For a strain gauge placed at 45° with respect to the axis of the shaft, the strain ε (m/m) measured by the gauge is:

$$\varepsilon = S\,(1 + \nu) \,/\, E \qquad (4)$$

Also

$$G = \frac{E}{2\,(1 + \nu)} \qquad (5)$$

where

G = Shear Modulus (Pa)
ν = Poissons Ratio
E = Youngs Modulus of Elasticity (Pa)

Therefore,

$$\varepsilon = \frac{S}{2G} \qquad (6)$$

Substituting (6) into (3) yields an expression relating torque and the strain as measured by the strain gauge:

$$T = \frac{\pi \varepsilon G d^3}{8} \qquad (7)$$

For a rotating shaft, power can be calculated from torque as follows:

$$P = 2\pi NT \qquad (8)$$

where

P = Power, W
N = Rotational speed, rev/s
T = Torque, N.m

CALCULATION OF RUNNING TORQUE AND OUTPUT POWER FOR N_2 COMPRESSOR MOTOR

N_2 Compressor Motor Shaft Parameters

$d = 60.325 \times 10^{-3}$ m (2.375 in)

$G = 7.72 \times 10^{10} - 7.93 \times 10^{10}$ Pa (11.2 - 11.5 Mpsi)

N = 60 rev/s (3600 rpm) approx

Measured Strain

Under normal operating conditions, the strain measured on the shaft, using the full bridge configuration, was approximately 74.4 $\frac{\mu m}{m}$.

Calculation of Motor Running Torque and Output Power

For $G = 7.72 \times 10^{10}$ Pa

Using equation (7)

$$T = \frac{\pi \varepsilon G d^3}{8}$$

$$T = \frac{\pi\,(74.4 \times 10^{-6}\ \text{m/m})(7.72 \times 10^{10}\ \text{Pa})\,(60.325 \times 10^{-3}\ \text{m})^3}{8}$$

$$= 495.2\ \text{N.m}$$

From Equation (8):

$$P = 2\pi\,TN$$

$$= 2\pi\,(495.2\ \text{N.m})\,(60\ \text{rev/s})$$

$$= 186.7 \times 10^3\ \text{W}$$

The table below summarizes the running torque and motor output power for the possible range of Shear Modulus.

	$G = 7.72 \times 10^{10}$ Pa	$G = 7.93 \times 10^{10}$ Pa
Running Torque N.m (in.lb)	495.2 (4383)	508.7 (4502)
Motor Output Power kW (HP)	186.7 (250.3)	191.8 (257.1)

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